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A review and design study of blade testing systems for utility-scale wind turbines

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A R T I C L E I N F O

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Since the blades are one of the most critical components of a wind turbine, representative samples must be experimentally tested in order to ensure that the actual performance of the blades is consistent with their specifications. In particular, it must be demonstrated that the blade can withstand both the ultimate loads and the fatigue loads to which the blade is expected to be subjected during its design service life. In general, there are basically two types of blade testing: static testing and fatigue (or dynamic) testing. This paper includes a summary review of different utility-scale wind turbine blade testing methods and the initial design study of a novel concept for tri-axial testing of large wind turbine blades. This new design is based on a blade testing method that excites the blade in flap-wise and edgewise direction simultaneously. The flap motion of the blade is caused by a dual-axis blade resonance excitation system (BREX). Edgewise motion is delivered by the use of two inclined hydraulic actuators and linear guide rail system is used to move the inclined actuators in the flap-wise direction along the blade motion. The hydraulic system and linear guide rail requirements are analyzed and an initial cost estimate of the proposed system is presented. Recommendations for future work on this proposed system are given in the final section of this work.

Contents

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1. Introduction/background

Presently, most utility-scale wind turbines have a horizontally mounted hub with three (or sometimes two) blades. As the blades become longer to capture more power, the static and dynamic loads on the blades and other components have increased. For example, a blade for a 1.5-MW turbine may be 34 m in length or greater and may weigh as much as 6000 kg (13,200 lbs) [1]. Because the blades are among the most critical components of the wind turbine, they have to be tested in order to ensure that their specifications are consistent with the actual performance of the blade. According to the International Electrotechnical Commission's blade testing standard, IEC 61400-23 [2]. The fundamental purpose of a wind turbine blade test is to demonstrate to a reasonable level of certainty that a blade type, when manufactured according to a certain set of specifications, has the prescribed reliability with reference to specific limit states, or, more precisely, to verify that the specified limit states are not reached and the blades therefore possess the strength and service life provided for in the design. Furthermore, it must be demonstrated that the blade can withstand both the ultimate loads and the fatigue loads to which the blade is expected to be subjected during its designed service life.

1.1. Blade testing methods

In general, blade testing methods fall into two main categories: fatigue (or dynamic) and static testing. The test load can either be load-based or strength-based. The purpose of the load-based test is to show that the blade will sustain the intended loads without failure. This type of test is normally used as part of a certification process. Strength based testing uses as-manufactured blade strength data as its basis, and the blade is tested to failure. This allows a direct verification of the blade strength and failure mechanism, and an assessment of ways in which the design computations, and the resulting design itself, might be improved. This method can be used to find the lowest strength location, relative to expected strength, within a broad region.

1.1.1. Static testing

In static testing, loads are applied to the blade statically in one direction to establish its ultimate strength. Non-destructive testing is done with the purpose of verifying a blade's ability to withstand extreme loads such as those caused by destructive very high but very rare wind speeds or by unusual transient conditions. It is carried out in order to determine the ultimate strength and failure mechanism of the blade under large static loads.

Static testing is accomplished in a number of ways. The most common of these uses an electric winch system, due to ease of control. Hydraulic actuators have also been used in the past but the large displacements required for longer blades make them an expensive option. Another way of performing a static test is to hang ballast weights from the blade at specified locations. In case of larger blades, the blade is attached to the test stand at an angle in order to prevent the tip of the blade from touching the ground, as shown in Fig. 1 [3].

1.1.2. Fatigue testing

This type of test is mainly used to identify structural defects inherent in either the design or manufacturing process. Fatigue tests are performed to verify the durability of the blade, when subjected to a cyclically varying loading profile. Fatigue tests apply a loading spectrum which may contain a 1 million to 5 million load cycles. They are typically performed in two primary directions, flap-wise and edgewise (lead–lag). The magnitude of the static loading is always higher than the fatigue loading. Tests to account for the multiaxial fatigue loads are often performed sequentially, for example, first in the edgewise direction followed by loading in the flap-wise direction. Dual-axis testing is another approach. Here, both flap and lead–lag loads are applied simultaneously. Dual-axis testing can, in principle, better simulate loads experienced in the field and can result in shorter overall test duration.

Currently, there are two methods used to apply these loads to the blade; these are generally referred to as forced displacement and resonant oscillation testing. Forced displacement testing uses long-stroke actuators or bell cranks and push rods to force the blade to a prescribed displacement that varies in time in a prescribed way. This is done in a cyclic manner and has the benefit of being able to apply nearly any combination or sequence of loading cycles to the blade. In general this type of loading works well for edgewise testing where the loads are closer to fully reversed bending than in the flap-wise direction. In the flap-wise direction, however, forced displacement testing requires very long stroke actuators and high forces. This would require a very large flow rate for the hydraulics that supply the load. Resonant testing uses an oscillating mass driven by an actuator attached to the blade through a frame. The mass is driven at the flap-wise resonant frequency. The resulting response of the blade provides the large displacement required with greatly reduced force and power.

There are few laboratories throughout the world that have the facility to perform static and fatigue testing of the wind turbine blades; Risø National Laboratories in Denmark, the Center for Renewable energy and Sources (CRES) in Greece, the Wind Turbine Materials and Constructions Knowledge Center (WMC) at TU Delft in Netherlands, the National Renewable energy Laboratories (NREL) in US, New and the Renewable Energy Center (NaREC) in United Kingdom and the LM Glasfiber in-house testing facility located in Lunderskov, Denmark. In the United States, large blade test facilities include the Massachusetts Wind Technology Testing Center (WTTC) in Charlestown, MA and the Large Blade Test Facility in Houston, TX (still under planning construction). Each of these test facilities features independently developed blade testing methods.



Fig. 1. Static testing using ballast weights and winches.



Fig. 2. Dual-axis forced-displacement test system.



Fig. 3. NREL BREX dual axis resonance test system.

Risø performs fatigue tests by applying cyclical loads in either the flap or lead–lag direction using an electric motor that rotates an eccentric mass. This testing method is referred to as the single-axis resonance test. The single-axis resonance test applies each component independently in two separate tests, thus making it less accurate for predicting life of the blade because it does not simulate the combined loading conditions experienced in the field. However, single-axis testing has several advantages over dual-axis forced-displacement test. By adding masses to the blade, it is possible to match the bending moment distribution in that single axis more closely to the bending moments which would be experienced in service. Although the added masses lower the system's natural frequency, the cycle frequency remains higher than forced-displacement test. Dual axis testing provides combination of loads, and also eliminates the need for sequential testing.

NREL, CRES and WMC use hydraulic actuators that apply loads at a single span-wise station on the blade in both flap-wise and lead–lag directions. This testing technique is referred to as dual-axis forced-displacement method. This method employs a servo-hydraulic system with actuators to exercise the blade in flap and lead–lag directions, at frequencies well below the blade's first fundamental flap-wise natural frequency, as shown in Fig. 2 [1]. The main advantage of this system is that the bi-axial loading creates strain profiles that more accurately agree with the in-service operating conditions, as compared to single-axis tests.

While this method is more accurate, it has several drawbacks. The forced loading system requires large forces and displacements from the hydraulic actuators. As a result, new actuators have to be designed and built each time a larger blade is used. As the actuator size increases, the hydraulic pumping requirements also increase. Accordingly, substantial equipments costs are incurred when increasing the capability of testing larger blades [1].

As the blades continued to grow larger in size, a new method was required to be developed to test the blades, keeping the costs down and to allow wind industry to compete in the energy market. At NREL, this led to the development of dual-axis blade resonance excitation system [1]. In this testing method, a small hydraulic actuator is used to displace a specified mass to excite the blade at its natural frequency in the flap-wise direction, while a bell crank system is used to provide displacement in the lead–lag direction, as shown in Fig. 3.

This testing methodology has the advantage of reducing hydraulic forces in both directions and being a universal testing device for the flap-wise direction. The main drawback of this system is the bell crank mechanism, which applies a point load in the lead–lag direction using a hydraulic actuator. An improvement on this system is the dual-axis universal resonance excitation (UREX) test method [4]. In this method, the bell crank mechanism is replaced by independent hydraulic actuators and masses in the saddle device, which resonates the blade in both flap and lead–lag directions. This system was tested on a small scale and proved to be a valid test method. Future work and tests are currently underway to refine and scale the system to provide a universal mechanism that can be used for any size blade.

As will be described in the following sections, a key purpose of this work was the development and design of a dual-axis blade testing method for larger wind turbine blades.

2. Overview of dual-axis testing of large wind turbine blades

2.1. Motivation

As previously discussed, loading of the blades can be done sequentially via single axis testing, e.g., first in the edgewise direction followed by testing in the flap-wise direction. Dual axis testing is another method of testing blades. In this case, both the flap and edgewise loads are applied simultaneously. This type of approach for testing the blades is preferred over single axis testing because it simulates the actual blade loads experienced in the field by including the phase angle between flap-wise and edgewise loads. Moreover, dual axis testing results in a shorter overall test duration.

2.2. Bell crank systems and their limitations

A schematic of a forced displacement test using a bell crank system is shown in Fig. 4. Ideally, a bell crank system should impart the force only in edgewise direction even when the flap-wise deflection is occurring. However, the system imparts an additional force as discussed next.

2.2.1. Cross-coupling of flap-wise and edgewise force components

As shown in Fig. 5, cross coupling is the effect of flap-wise force being introduced due to edgewise actuator, or edgewise load component introduced by the flap-wise actuator. This cross-coupling requires correction factors to be incorporated into the control system for the testing mechanism.

2.2.2. Induced pitch moments

As shown in Fig. 4, the push rod is connected to the blade on the pitch axis. Because the blade cannot be cut, the push rod must be attached to the front of the blade. In this case, the flap-wise component of the push rod force creates an undesirable pitching moment. If one is concerned with floor space requirements for a large blade test facility, every effort has to be made to keep the push rod length as short as possible. On the other hand, a short push rod results in larger push rod angles and a larger flap-wise component, thereby

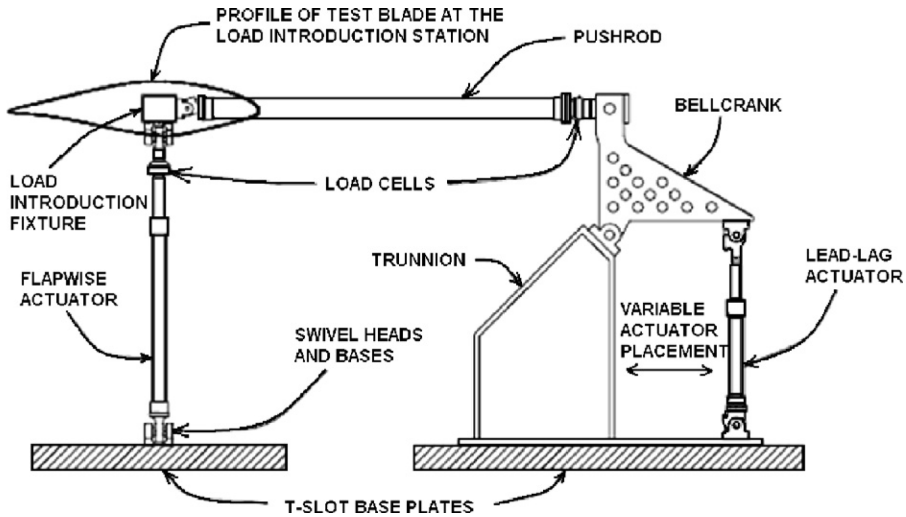


Fig. 4. Schematic of forced displacement test using a bell crank system.

exacerbating the pitch moments and torsion about the pitch axis. These undesired pitch moments and deflections might result in unrealistic load conditions thus not simulating the actual load conditions. Also, a short push rod will result in flap-wise deflection at an undesired frequency resulting in non-sinusoidal waveform. For these reasons, the push rod has to be made longer. Here, building and cost constraints come into play and force a compromise solution.

In order to formulate the push rod sizing, one needs to know the acceptable pitch moment. This may be difficult to determine because it which is expected that the allowable moment will vary between blade manufacturers and blade designs. One approach for sizing the push rod length that could be considered a reasonable compromise is to size the push rod such that the undesirable flap-wise force component is less than 10% of the total push rod force. An estimation approach for push rod length used by NREL [5], assumed a simplified bell crank geometry with a push rod initial height aligned with the blade deflection. Results from this work are shown in Table 1.

In general, the length of the push rod must be approximately 5 times the flap-wise deflection at the 70% station in order to meet the 10% constraint on the vertical push rod force component.

2.2.3. Push rod sizing

The length of the push rod required to maintain a flap-wise force of less than 10% of the push rod force for a 70 m blade is 30 m. A 30 m long push rod subject to 37 tonnes of axial force must be very large and heavy to avoid buckling. To avoid Euler buckling with a safety factor of 4.0, the push rod for this blade will be approximately 0.45 m (18″) in diameter with a 0.019 m (0.75″) wall and weigh 6 metric tons. The weight of the pushrod induces an additional static force downwards and an additional moment in the piston direction, while the large mass of the push rod further increases the actuator forces. In addition, such a long push rod would interfere with testing in the two adjacent bays. One way to reduce the length, weight, and cost of such a long push rod is to relax the constraint to maintain a flap-wise push rod force component less than 10% of the push rod force.

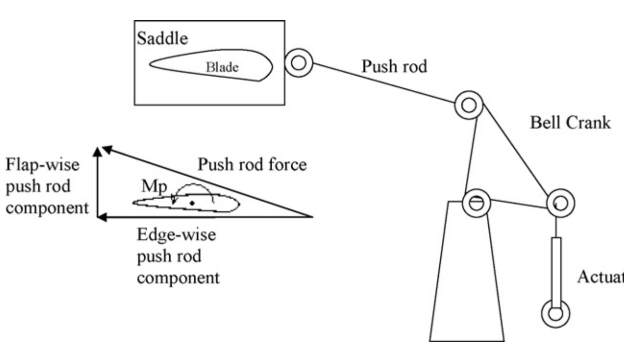


Fig. 5. Schematic of bell crank geometry.

Table 2 shows the reduced push rod requirements if the constraint is relaxed to 20%. In this case, the push rod length could be reduced in half and diameter could be reduced to (0.308 m) 12″ with the same wall thickness thereby reducing the mass to 2 metric tons. However, the induced pitch moment and increased coupling induced by the nearly doubled flap-wise push rod load component (5 tonnes) may not be acceptable to the customer.

2.2.4. Bell crank spanwise positioning

One alternative to reducing the space and mass requirements of a bell crank is to place the bell crank closer to the root where the flap-wise deflections are smaller. Positioning the bell crank closer

Table 1
Push rod length and force calculations.

Blade length (m)	50	60	70	70
Phase angle (deg)	90	90	90	70
Flap deflection (m) (2× amplitude)	3.5	4.5	6.0	6.0
Edge deflection (m) (2× amplitude)	0.5	0.7	0.9	0.9
Push rod length (m)	18	23	30	30
Max push rod force (metric tons)	13	23	37	37
Max flap-wise push rod component (metric tons)	0.7	1.1	1.8	2.5

Table 2
Deflections, push rod length, and force components required to maintain a push rod force component that is less than 20% of the push rod force.

Blade length (m)	50	60	70	70
Phase angle (deg)	90	90	90	70
Flap deflection (m) (2× amplitude)	3.5	4.5	6.0	6.0
Edge deflection (m) (2× amplitude)	0.5	0.7	0.9	0.9
Push rod length (m)	9	11	15	15
Max push rod force (metric tons)	13	23	37	37
Max flap-wise push rod component (metric tons)	1.3	2.3	3.6	4.9

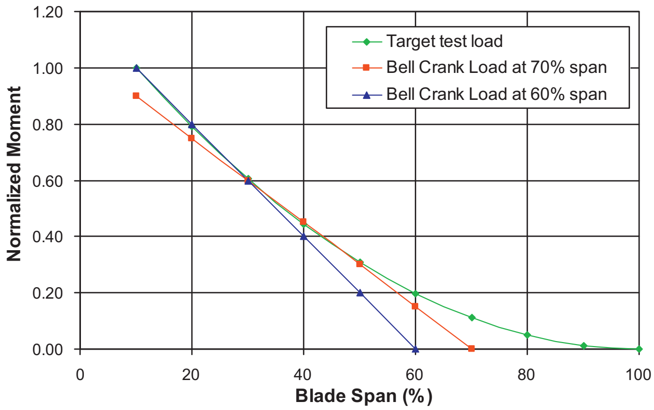


Figure 6. Normalized target and bell crank moment distributions for an edge-wise fatigue test.

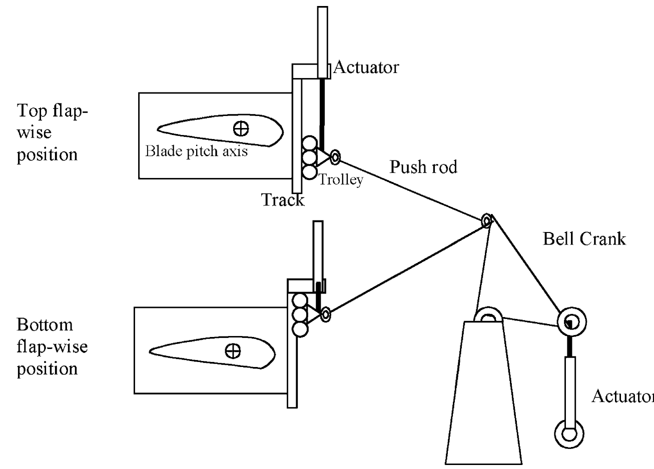


Fig. 7. Schematic of an actively positioned bell crank system.

to the root, however, alters the targeted moment distribution of the test. The area of interest in a fatigue test is approximately 20–50% of the blade span. All calculations in this report assume the bell crank is positioned at 70% span. By positioning the bell crank at approximately 70% span location, a reasonable approximation of the target edgewise bending moment distribution can be obtained, except at the tip, where the moment is small (see Fig. 6) [5].

Positioning the bell crank closer to the root (i.e. 60% span) better matches the target test load inboard but will insufficiently load the outboard sections of interest. In addition, the push rod must apply more force when positioned inboard but at a smaller displacement. Positioning the bell crank further outboard will have the opposite effects. Multiple edgewise actuators would result in a closer match to the target moment distribution, but will increase the complexity of the test substantially as coordinating the forced displacement edgewise deflections of the actuators is expected to be challenging.

3. Review of edge-actuator designs

Alternative bell crank designs may reduce the space and cost of the traditional bell crank system. In addition, an alternative design may facilitate tri-axial testing of wind turbine blades by enabling the control of pitch degree of freedom. In this chapter, several bell crank system configurations have been considered. This section reviews the two of the most promising designs: An actively positioned bell crank (ABC), and a blade-mounted actuator.

3.1. Actively positioned bell crank

An actively positioned bell crank (Fig. 7) could possibly eliminate the problem caused by induced pitch moments and possibly reduce the amount of span-wise and edge-wise coupling. An actively positioned bell crank uses a second actuator to actively position a trolley to control the amount of pitch induced into the blade. If it is desired to minimize the pitch induced into the blade, the trolley is positioned to align the push rod with the pitch axis. Additionally the flap-edge coupling could be slightly reduced because the motion of the trolley could be used to reduce the inclination angle of the push rod. By reducing the pitch moment and coupling forces, a shorter, lighter push rod can be used in the system. In addition, active control of the pitch moment could facilitate more accurate simulation of the operating conditions observed in the field by allowing tri-axial testing (flap-wise, edge-wise, and pitch).

As noted by [6], the actively positioned bell crank (ABC) requires a moderate amount of development that includes system modeling, design work, fabrication, and testing on a small to medium sized blade. The configuration described in this paper shares most of its structure with the proven NREL bell crank system so the latter could be used for prototyping.

3.2. Blade-mounted edgewise actuator concepts

The blade mounted edge actuator system displayed in Fig. 8 was considered by NREL and its CRADA partner NaREC in 2005 [6]. The system uses an actuator mounted on the blade and a trolley to maintain a horizontal edgewise force. This system minimizes the coupling and dramatically reduces the amount of building space required for dual-axis testing by replacing the push rod with an actuator. There is still significant pitch excitation, however, because the actuator is offset from the pitch axis. Furthermore, rigidly mounting the actuator to the blade saddle results in bending moments being applied to the actuator piston. This results from the saddle rotation about the test stand's horizontal and vertical axes. These bending moments are likely to damage the actuator and apply undesirable moments to the blade saddle.

An improvement to NaREC's Blade-Mounted Edgewise Actuator Concept is to use two edgewise actuators on the top and bottom of the blade (see Fig. 9, Embodiment 1). Using two actuators symmetrically positioned about the pitch axis dramatically reduces or eliminates the pitch moment and can even facilitate active control of the pitch moment for tri-axial testing (deflection in the flap, edge, and pitch directions). A second benefit is that each actuator is mounted with a universal joint at each end, thereby eliminating the

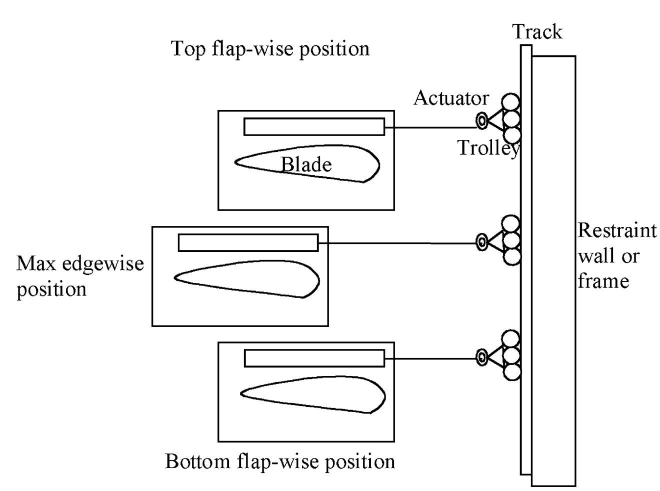


Fig. 8. Schematic of a blade-mounted edgewise excitation system concept.

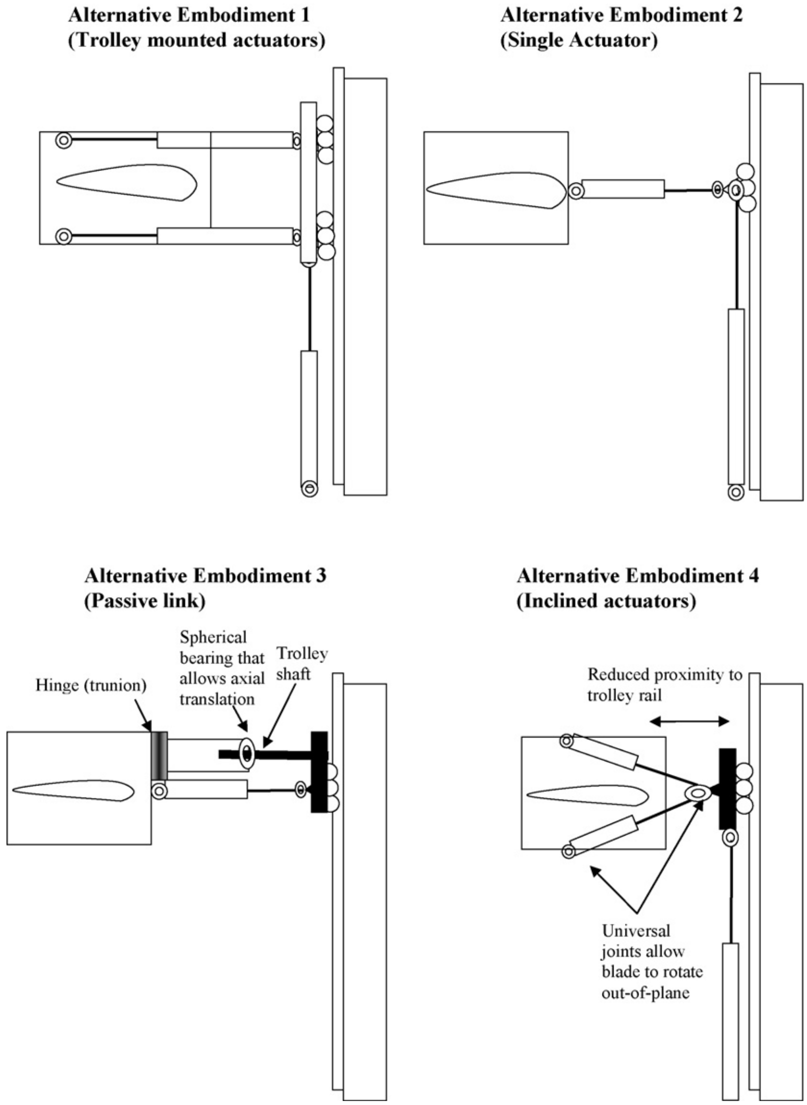


Fig. 9. Alternative embodiments of NREL's blade-mounted edgewise excitation system concepts.

bending forces due to the rotations of the saddle. A third benefit is that using two actuators reduces the size of the actuator. Horizontal mounting of very heavy (approximately 356–445 kN) actuators is believed to result in premature damage to the actuator seals and bearings [7]. The trolley's vertical position must be actively controlled using some sort of trolley positioner. Otherwise, the system behaves like a four-bar-linkage and the trolley will not stay aligned with the blade. The vertical control of the trolley could be achieved by adding a motor to the trolley or by adding a long stroke actuator as in Fig. 9 (Embodiment 1). Trolley mounted actuators will slightly reduce the mass mounted on the blade and provide an alternative means of routing hydraulic lines.

A second alternative (Embodiment 2), as can be seen in Fig. 9, is to use a single actuator to reduce the complexity of the system by eliminating one of the actuators. However, if tri-axial testing is desired, this solution significantly complicates the control system and would result in coupling of the flap-wise and edgewise loads. In addition, a single actuator will be significantly more massive and will have to be sized to apply the entire edgewise force. This large actuator could be more sensitive to horizontal mounting.

A third alternative is to use a passive trolley positioning system to simplify the system and reduce the shear loads on the edgewise actuators (Embodiment 3). In this alternative, the complexity is reduced by eliminating the need to actively control the vertical position at the expense of adding a passive positioner that may be difficult to design to allow all the desired degrees of freedom.

A fourth alternative (Embodiment 4) uses a passive trolley system with two inclined edgewise actuators on the top and bottom of the blade, mounted via universal joints or other configurations that result in similar degrees of freedom. The inclined orientation converts a large portion of actuator bending load to actuator axial loads, thereby increasing seal life and service interval. Furthermore, the use of two actuators tends to be more forgiving of horizontal or near-horizontal positioning. The inclined actuator system is lighter and more easily controlled than the other embodiments. This facilitates the possible use of multiple actuator systems along the span of the blade. Perhaps most importantly, using two inclined actuators allows the blade to be significantly closer to the trolley rail, proportionally reducing pitch moments imparted by the system mass and trolley friction.

Embodiment 4 has many advantages over the other alternatives, and, as discussed in the next section, was considered for a more detailed analysis.

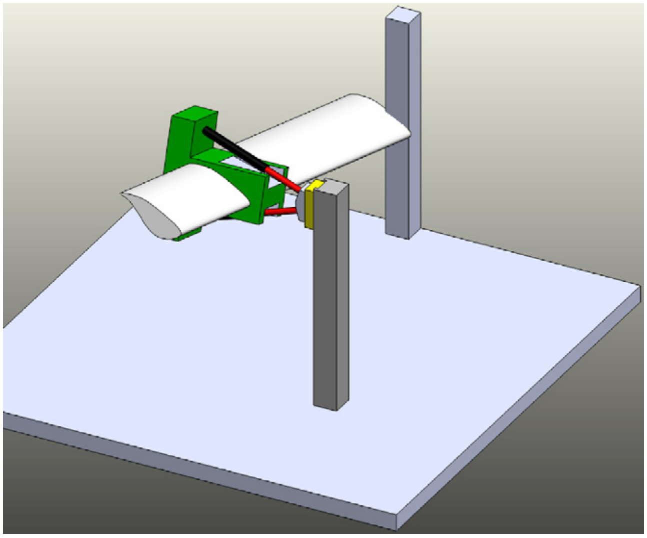


Fig. 10. Design of model in SolidWorks.

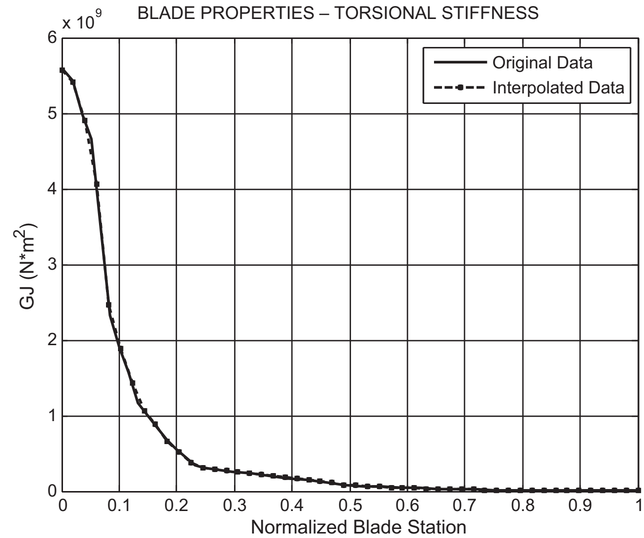


Fig. 11. Torsional stiffness along the length of the blade.

4. Case study design of selected system

4.1. Overview

In this section a case study design of a representative system will be presented. Before proceeding to the dynamics and feasibility of the mechanics of the overall design, the system was modeled using 3-D modeling software that focused on the kinematics of the design. A simplified model, as shown in Fig. 10, was constructed using SolidWorks [8].

For the next step, a preliminary design of the system was required to calculate the dynamics involved and the feasibility of the design. For example, one needs to know the hydraulic system and linear guide rail design requirements.

4.2. Summary of analysis procedure

The design requirements and calculations were made taking a specific blade into consideration. The input data for the blade was generated using the software code FAST for a 5 MW, 62 m blade [9]. As described in [7], the design process required the calculation of the following blade properties:

(1) Mass per unit length
(2) Chord length
(3) Flap stiffness
(4) Edge stiffness
(5) Axial stiffness
(6) Torsional stiffness

In all cases, the blade data generated using FAST were interpolated according to the required normalized blade sections. The original data and the interpolated data matched very closely (e.g., Fig. 11 presents the torsional stiffness blade properties used for this study).

4.3. Results of calculations

The analysis of the selected blade, with and without saddles, with the test configuration as shown in Fig. 9 (Alternative embodiment 4) was carried out using a MATLAB code generated at NREL [10]. The source code obtained the blade properties, target loads, and saddle specifications from an input Excel file. The input data was then distributed into a finite element blade model and run through the appropriate test simulation. Source code features include the ability to generate missing properties and loads using curve fits based on blade length, as well as built-in optimization routines to determine locations and loads of saddles. Once the target load was determined, the applied load was calculated by combining the moments of several loading points to get a distributed load. Three different cases were considered:

1. Static case 1 – stationary blade without saddles, under its own weight
2. Static case 2 – stationary blade with saddles
3. Dynamic case – a blade moving in the flap-wise directions with saddles on it

The full results of these studies are presented in [7]. As a representative example, the results for the dynamic case are summarized next.

4.3.1. Results for the dynamic case

The dynamic analysis was carried out with saddles attached to the blade at three different position so as to match the bending moment distribution along the length of the blade. A blade resonant excitation system (BREX) was used for flapping the blade at the resonant frequency in the flap-wise direction and inclined actuators impart the desired force in the edgewise direction. The calculational results are shown in Table 3 and Fig. 12 presents the flap and edge moments.

4.4. Summary of system hardware requirements

As detailed in [7], in order to arrive at a capital cost estimate for this example design, a series of calculations for the system hardware requirements had to be carried out. The following section summarizes the results of this work for the key components of the system.

Table 3
Dynamic analysis calculations and results.

Flap tip deflection	4.36 m	14.30 ft
Flap root alt moment	5973 kNm	4406 kipft
Edge tip deflection	1.70 m	5.57 ft
Edge bending root moment	10469 kNm	7723 kipft

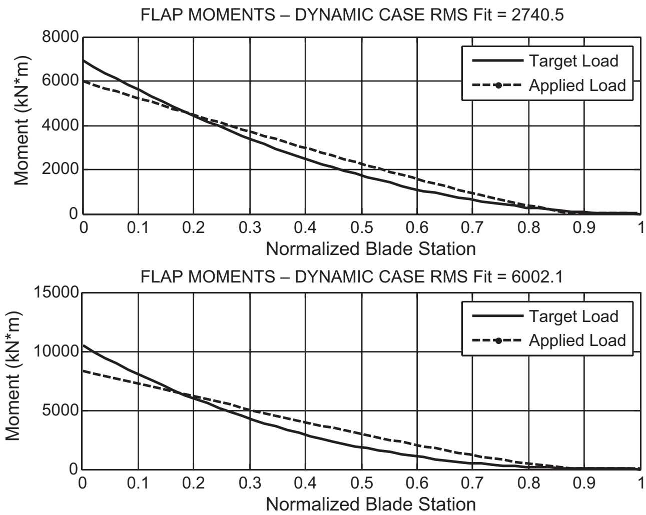


Fig. 12. Flap and edge moments for dynamic case.

4.4.1. *Hydraulic actuator*

Initially, the hydraulic force required to be delivered by one actuator must be calculated. The normal operating pressure for the hydraulic cylinders used for blade testing is 20.7 MPa (3000 psi). If the guide rail assembly is allowed to move on its own, the BREX system has to apply much greater force taking the weight of the hydraulic actuators, universal joints and guide rails assembly into consideration. This is resolved by attaching another hydraulic actuator that moves the guide rail assembly up and down simultaneously with the flap-wise motion of the blade, taking the weight of hydraulic actuators and linear guide rail assembly.

As calculated for this design case, the edge root bending moment at 70% blade station is equal to 10,469 kNm

$$\text{Thus, the total force required} = \frac{\text{Edge root bending moment}}{\text{Length of the blade at 70\% station}} = \frac{10469}{0.7 \times 61.33} = 243.87 \quad \text{kN}$$

Since this force is being delivered by two actuators inclined at an angle θ, the force delivered by one actuator = (total force required/$2 \times \cos\theta$)

Stroke length required for an actuator = $2 \times$ deflection at half amplitude/$\cos\theta$

Since pressure = force/area, the cylinder diameter, $d = 2 \times \sqrt{\text{force}/(\text{pressure} \times \pi)}$

Based on the above equations, and with a hydraulic pressure of 3000 psi, the specifications for the hydraulic cylinder are summarized in Table 4.

As detailed in [7] and summarized below, a cost estimate for the hydraulic components was obtained from the MTS company.

Table 4
Hydraulic system requirements.

Angle between actuators (degrees)	30	40
Force to be delivered by one actuator (kN)	126 (28.38 kip)	130 (29.17 kip)
Edge frequency (Hz)	0.62	0.62
Edge deflection half amplitude at 70% (m)	0.8	0.8
Stroke length required (m)	1.65	1.7
Cylinder diameter (in.)	3.47	3.52
Flow rate (gpm)	192	197

Table 5
Preliminary capital cost estimate for system.

1.	Qty 2, MTS series 201 hydraulic actuator -35 kip, 70 in. stroke actuators with 400 gpm servo valves -Used for the edgewise displacement of the blade–MTS	$250,000
2.	Qty 3, 180 gpm hydraulic power units	$550,000
3.	Multi channel control system	$150,000
4.	Hydraulic distribution	$100,000
5.	Custom designed test fixturing	$200,000
6.	Installation support -Provided by MTS Corporation	$25,000
7.	MTS custom built hydraulic actuator with 3 m stroke length -Used for imparting vertical motion to the guide rail assembly	$150,000
8.	Linear guide rail system – size 55 long style carriages (provided by Schaeffler) -Attached to the saddle and used for carrying the saddle and 2 hydraulic actuators assembly to follow the flap-wise movement of the blade	$6000
	Total cost	$1,431,000

4.4.2. *Linear guide rail system*

The linear guide rail system is attached to the saddle on the blade at the 70% station where the flap deflection was calculated to be 2.82 m. The first flap frequency is 0.365 Hz. The motion of the flap can be described as a sinusoidal wave with amplitude of 2.82 m and a frequency of 0.365 Hz, for which the equation of motion is:

$$X = A \sin(2\pi f t),$$

where X is displacement of the blade at 70% station in the vertical direction, A is amplitude, or flap deflection; f is frequency, t is time (in s).

As summarized in [7], for these requirements, the Schaeffler Company provided the relevant product information and a price quote. They recommended four-row linear recirculating ball bearing and guideway assemblies. Specifically, based on their recommendations, a size 55, long style carriage was the most appropriate choice to use for this application.

Also, the vertical post to floor, also referred to as carriage, is an I-beam, available in numerous variants. It has saddle plates with hardened and precision ground rolling element raceways. The slider or guideway is made from hardened steel and is ground on all the faces.

The guide rail system needs another means by which it can move up and down along with the flap-wise motion of the blade, which can be achieved by attaching another hydraulic actuator for this purpose. The maximum force acting vertically against which the hydraulic actuator has to work is the weight of two inclined actuators, the block, the weight of the linear guide rail system, the force of friction acting between the four row linear bearing assembly and the maximum acceleration in the whole system. This force comes out to be 18.3 kN and to serve the purpose, we need a hydraulic actuator with a ∼3 m stroke length. MTS recommended the use of hydraulic actuator with 25 kN force rating for the purpose, delivering a stroke of 3 m at a frequency of 0.365 Hz. The flow rate required here is 50 gpm and the cost of the actuator is $150,000 approximately.

4.4.3. *Summary of estimated system costs*

Based on the previous design estimates and other minor components (see [7]), Table 5 presents a first order estimate of the capital costs for this proposed system.

5. Conclusions/recommendations

Hybrid testing and forced-displacement testing of wind turbines blades in the edgewise direction require a means of forcing the

blade displacement in that direction. During the past 10 years of testing, NREL has used a bell crank system to impart this displacement. However, NREL's experience with the NREL bell crank system is limited to blades less than 40 m long. It is expected that customers will request that dual axis testing in some form be performed at the large blade test facilities on larger blades. The conventional bell crank systems previously used by NREL to perform dual-axis testing are likely to be expensive due to the lateral space requirements (push rod length) and system mass required to sufficiently mitigate the flap/edge coupling and induced pitch moment.

As summarized in this paper, one alternative design is an actively positioned bell crank system (ABC). The alternative design which is described in this paper has been designed in SolidWorks to confirm the kinematics of the model. In addition, the system requirements including hydraulic system, linear guide rail system, flange joint, universal joint specifications have described. Although this concept addresses the induced pitch problem, an ABC may not sufficiently reduce the lateral space required for a bell crank system. However, using a passive trolley system with two inclined edgewise actuators, mounted via universal joints allows the blade to be significantly closer to the trolley rail, proportionally reducing pitch moments imparted by the system mass and trolley friction.

A hydraulic system configured for this application would require actuators, hydraulic power units (HPU), control system, hydraulic distribution, fixturing and engineering support which will cost approximately $1.4 million. It also features a linear guide rail system would use four-row linear recirculating ball bearing and guideway assemblies. The cost for building this system is more than the systems being used today. On the other hand, it is a more efficient and better way to test large wind turbine blades. Instead of testing the blade in flap-wise and edgewise direction separately for months, this design is capable of testing the blades in both directions at the same time. This will reduce the testing time by 50%.

The blade-mounted edgewise excitation system requires dramatically less space and can potentially eliminate the flap/edge coupling and pitch problems, but significant development challenges remain. One challenge is that a trolley bearing system must be identified or developed capable of very high loads and relatively fast speeds (averaging up to 6 m/s or 13 mph) continuously reversed having a displacement of 3 m. A second challenge is that a control system must be developed for the actuation systems that ensures the trolley stays vertically aligned with the blade (to avoid flap/edge coupling) and imparts the desired pitch moment. A third challenge is the complication involved with custom engineered hydraulic actuator with a 3 m long actuator stroke. Finally, other unexpected challenges may arise during implementation of this approach. For example, the simplified schematics may not fully address how factors such out-of-plane loads will affect the saddle attachment to the blade. In summary, future work should include designing a prototype for this concept.

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